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CORRELATION OF EXHAUST-VALVE TEMPERATURES WITH ENGINE OPERATING CONDITIONS AND VALVE DESIGN

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SUMMARY

The theory correlating engine-cooling variables developed in a previous report was used as a basis for the development of a semi-empirical equation to correlate exhaust-valve temperature with engine conditions. A term is included in the equation that is a measure of the thermal resistance of the heat-flow path between the crown of the exhaust valve and a point on the outside surface of the cylinder head. A means for comparing exhaust valves of different designs with respect to cooling is consequently provided. The necessary empirical constants included in the equation were determined from engine tests of a large air-cooled cylinder.

INTRODUCTION

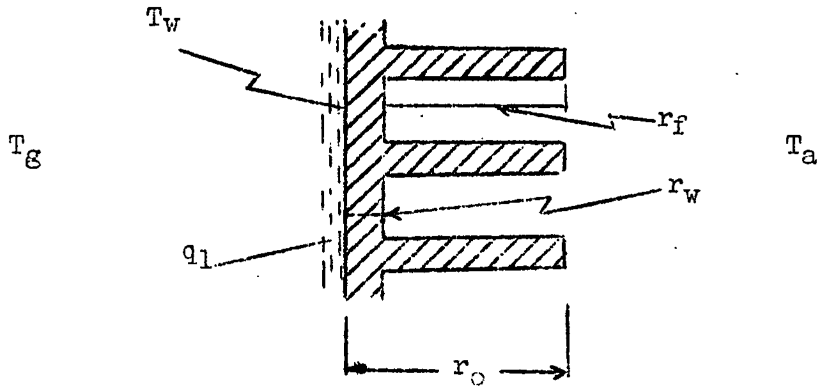
A number of investigations have been conducted on the operating temperature of exhaust valves. As early as 1923, Gibson and Baker (reference 1) measured the temperature of a hollow-stem exhaust valve without internal coolant by means of a thermocouple and found that the operating temperature of the exhaust valve in a cylinder of low specific output (less than 0.25 hp/cu in.) varied between 600° C and 750° C, depending mainly on fuel-air ratio, cylinder cooling, and spark timing. Reference 1 also states that under abnormal conditions such as preignition, the valve temperature might exceed 800° C. Subsequent developments in valve and cylinder design have permitted much higher specific outputs than 0.25 horsepower per cubic inch with approximately the same range of exhaust-valve temperatures. The operating temperature and the effects of several operating variables on the temperature of a sodium-cooled exhaust valve in current use are shown in reference 2 and an indication of the extent to which valve temperatures can be influenced by valve design is reported in reference 3. A review of available data on exhaust-valve temperatures, however, has shown no attempt to develop a mathematical expression relating design and operating variables to exhaust-valve temperature.

Pinkel (reference 4) has shown that average head and barrel temperatures can be mathematically correlated with engine and cooling variables. An analysis parallel to that developed by Pinkel is used herein to set up semiempirical equations correlating exhaust-valve temperature with engine operating conditions and valve design. This analysis differs from that of reference 4 in that treatment is limited to the portion of the cylinder head that conducts heat from the exhaust-valve crown to the cooling air. The thermal resistance between the valve crown and the cooling air is included in the analysis. The necessary experimental constants have been determined from engine tests conducted during 1944 at the NACA Cleveland laboratory on a conventional air-cooled cylinder.

Inasmuch as one of the principal purposes of cylinder cooling is to prevent overheating and subsequent failure of critical cylinder parts, this type of correlation in which the temperature of a particular point within the combustion chamber is related to engine variables is valuable in determining cylinder-cooling requirements. The correlating equation is developed herein to determine the manner in which each of the engine and cooling variables affects exhaust-valve temperature and to provide a means of predicting valve-operating temperatures with a minimum of testing. The same equation, however, with proper selection of empirical constants can be applied to the spark-plug electrode or any other hot spot within the cylinder combustion chamber.

DERIVATION OF CORRELATING EQUATION

The problem of relating the exhaust-valve temperature to the numerous operating conditions and design features of the cylinder is most easily attacked by computing a balance of the heat transferred to and from the valve. The heat transferred to the valve is influenced by only the film coefficient of heat transfer between the working fluid and the valve and the temperature difference between them. Heat transferred away from the valve crown is a function of only the thermal resistance between the crown of the valve and the cooling air at the exterior of the cylinder and the temperature difference between the valve crown and cooling air. These parameters can be readily demonstrated in the following sketch for the simple case of a point on the inner wall of the combustion chamber:



where

T_a cooling-air temperature

T_g mean effective gas temperature

T_w wall temperature

q_1 coefficient of heat transfer from gas to wall

r_f thermal resistance of fins

r_o over-all thermal resistance from point on inside of cylinder to cooling air

r_w thermal resistance of cylinder wall

When a heat balance is set up for a point on the inside surface of the wall, the heat transferred to the wall H is

$$H = q_1 A (T_g - T_w) \tag{1}$$

where A is the area considered. Heat transferred away from the wall H is

$$\begin{aligned} H &= \frac{1}{r_o} A (T_w - T_a) \\ &= K_o A (T_w - T_a) \end{aligned} \tag{2}$$

where K_o is the over-all conductivity from a point on the inside of the wall to the cooling air.

The elimination of H in equations (1) and (2) gives the following equation relating the temperature of a point on the cylinder wall to the surrounding thermal conditions:

$$\frac{T_w - T_a}{T_g - T_w} = \frac{q_1}{K_o} \qquad (3)$$

It is now evident that most operating and design conditions may be related to T_w through analysis of their effects on T_g, q_1, r_o, and T_a.

The evaluation of these parameters for the exhaust valve is accomplished by expressing r_o in terms of several component parts as shown in figure 1.

The following symbols are used in figure 1:

T_v exhaust-valve temperature

r_b, r_f, r_g, r_v, and r_w thermal resistances between points

In this case, heat is removed from the valve crown through two parallel paths: one through the valve stem, having a resistance of $r_v + r_b + r_f$; and the other through the valve seat having a resistance of $r_w + r_g + r_f$. The effective thermal resistance of the two paths in parallel is given by the relation

$$\begin{aligned} \frac{1}{r_o} &= \frac{1}{r_v + r_b + r_f} + \frac{1}{r_w + r_g + r_f} \\ &= K_o \end{aligned} \qquad (4)$$

In addition to the heat added to the valve crown, heat is also transferred to the under side of the valve head and the valve-guide boss from the hot exhaust gases flowing through the exhaust port. This additional heat flow through the valve and valve-guide boss and the additional heat-flow paths it would follow would therefore have to be considered in any exact analysis. Consideration of the additional heat flow and heat-flow paths in a semiempirical analysis as presented herein, however, would make the final equation too complex for practical use. Moreover, experimental data presented in the discussion indicate that the effects of this additional heat flow on exhaust-valve temperature can be treated as part of the thermal-resistance factor without an appreciable loss in accuracy of calculated valve temperatures.

If a heat balance for the exhaust-valve crown similar to that for the point on the inside wall of the cylinder is set up, the following equation is obtained:

$$\frac{T_v - T_a}{T_g - T_v} = \frac{q_1}{K_o} \tag{5}$$

The solution is complete when T_g, q_1, and K_o are related to engine operating conditions and valve design.

The coefficient of heat transfer from the combustion gases to an exposed surface in the combustion chamber has been shown by Pinkel (reference 4) to be approximately an exponential function of the engine power as represented by the equation

$$q_1 = K_1 I^n \tag{6}$$

where

I indicated horsepower

K_1, n constants for a given cylinder

Inasmuch as experimental data present herein indicate that equation (6) represents with reasonable accuracy the coefficient of heat transfer from the gases to the exhaust valve, radiation effects will be assumed to be included in the coefficient.

The over-all conductivity between the exhaust-valve crown and the cooling air K_o can be approximately represented by the equation

$$K_o = \frac{1}{r_x + r_f} \tag{7}$$

where r_x is the equivalent thermal resistance between the valve crown and the outside of the cylinder. The value of the thermal resistance of the fins r_f may be represented by the relation (reference 4)

$$\frac{1}{r_f} = K_2 \, (\sigma \Delta p)^m \tag{8}$$

where $\sigma\Delta p$ is the cooling-air pressure drop across the cylinder and K_2 and m are constants depending upon the design of the fins and baffles.

Equation (5) can now be written

$$\frac{T_v - T_a}{T_g - T_v} = K_1 \, I^n \left[r_x + \frac{1}{K_2 \, (\sigma\Delta p)^m} \right]$$

$$= \frac{K_1}{K_2} \, I^n \left[K_2 \, r_x + \frac{1}{(\sigma\Delta p)^m} \right]$$

Let

$$K_2 \, r_x = C$$

and

$$K_1/K_2 = K$$

Then

$$\frac{T_v - T_a}{T_g - T_v} = KI^n \left[\frac{1}{(\sigma\Delta p)^m} + C \right] \tag{9}$$

where C is a thermal-resistance factor, the value of which is dependent upon the design of the exhaust valve and port.

Equation (9) may be used to relate valve temperature to most engine operating conditions and valve-design features. The most important of these variables except fuel-air ratio and combustion-air temperature appear directly in the equation. The relation between valve temperature and fuel-air ratio is a complex one. Consequently, an experimentally determined curve of T_g against fuel-air ratio for a given combustion-air temperature is used to select the proper value of T_g for use in equation (9).

APPARATUS AND TEST PROCEDURE

Tests were conducted to determine the constants to be used in equation (9) that are applicable to the particular air-cooled cylinder used in the sample calculations that follow. For this purpose, the test cylinder was mounted on an NACA universal test engine crankcase. The cylinder bore was $6\frac{1}{8}$ inches; the stroke used was 7 inches; and the compression ratio was 6.7. Standard laboratory equipment was used to measure engine speed, power output, cylinder temperatures, and fuel and air consumption. A diagrammatic sketch of the test engine and auxiliary laboratory equipment is shown in figure 2. Valve operating temperatures were measured by a constantan-valve steel thermocouple (fig. 3).

The thermal-resistance factor C was determined for three sodium-cooled valves of different design. A comparison of the design features of these valves is shown in figure 4 and the main differences are summarized in the following table:

Valve	Stem diameter (in.)	Throat diameter (in.)	Crown thickness (in.)	Crown coating
A	0.995	0.690	0.18 to 0.21	AMS 5682
B	.682	.283	.14 to .16	None
C	.682	.422	.14 to .16	None

Valve B was also tested without sodium to determine the effect of the coolant on the thermal-resistance factor of the valve.

In order to evaluate the constants to be used in equation (9) and to check the validity of the equation, the following four series of tests were made:

1. The cooling-air pressure drop across the cylinder was varied in the first series while other conditions of operation were held constant at the following values:

Engine speed, rpm 2200
Combustion-air temperature, ^{o}F 150
Fuel-air ratio 0.072
Spark timing, degrees B.T.C. . . . . $22\frac{1}{2}$

The purpose of these tests was to secure data needed in evaluating the constant m and thermal-resistance factor C and to determine the mean effective gas temperature T_g at a fuel-air ratio of 0.072.

In the determination of T_g it was also necessary that the temperature of a point on the outside surface of the cylinder nearest the exhaust-valve guide be known. The location of the thermocouple used to measure this temperature is shown in figure 1.

2. In the second series the power output of the engine was varied while all other conditions were kept constant at the following values:

Engine speed, rpm	2200
Combustion-air temperature, °F	150
Fuel-air ratio	0.072
Cooling-air pressure drop across cylinder, inches water	16
Spark timing, degrees B.T.C.	$22\frac{1}{2}$

The purpose of these tests was to evaluate the constants n and K.

3. In addition to the first two series of tests conducted in order to evaluate the constants for the correlating equation, a survey of valve operating temperatures over a range of engine conditions was made in the third series. Only one of the engine conditions listed below was varied at a time, while the others were held constant at the given values:

Indicated horsepower	117 and 71.5
Engine speed, rpm	2200
Fuel-air ratio	0.072
Spark timing, degrees B.T.C.	$22\frac{1}{2}$

4. In the fourth series of tests the temperature of each valve was determined at the following test conditions in order that a completely independent check on the accuracy of the equation might be made:

Engine speed, rpm	2300
Indicated mean effective pressure, pounds per square inch	215
Fuel-air ratio	0.09
Combustion-air temperature, °F	150
Rear-spark-plug-bushing temperature, °F	425

RESULTS AND DISCUSSION

Determination of Constants

All the necessary constants used in the correlating equation were evaluated for a given cylinder and valve combination from two engine tests; in the first test the cooling-air pressure drop across the cylinder was varied and in the second test the indicated horsepower was varied. From the data of the first test, the mean effective gas temperature T_g for the exhaust valve was estimated by plotting a curve of the temperature difference between the valve crown and a point on the outside surface of the cylinder head nearest the valve guide (fig. 1) against temperature of the outside surface of the cylinder. (See fig. 5.) The value of T_g is found be extrapolating the curve to the point where the temperature difference, and hence the heat transfer between the two points, is zero. A value of 2200° F at a fuel-air ratio of 0.072 for T_g was chosen as being most representative of the available data. The accuracy of the equation for correlating exhaust-valve temperatures, however, like those for correlating head and barrel temperatures is much more sensitive to the variation in T_g with engine operating conditions than to the initial temperature selected. Effective gas temperature T_g obtained in this manner is not intended to represent the true exhaust-gas temperature but should be used only as a means of correlating valve-cooling data.

The value of the constant m was obtained from the slope of a logarithmic plot of $\frac{T_o - T_a}{T_g - T_o}$ against cooling-air pressure drop $\sigma\Delta p$ (fig. 6) where T_o is the temperature of the point on the outside surface of the cylinder head nearest the exhaust-valve guide. The thermal-resistance factor C was determined by using the data of the first test and simultaneously substituting the values obtained for two test conditions in equation (9). Inasmuch as large variations in external cooling have relatively little influence on exhaust-valve temperature, small changes in any of the other operating variables (for example, fuel-air ratio) may offset large changes in cooling-air flow. It is therefore necessary that the test be carefully controlled and the test cover as wide a range of cooling-air flow as possible in order to get a representative value of C.

The remaining constants n and K were evaluated from the second test, n from the slope of a logarithmic plot of $\frac{T_v - T_a}{T_g - T_v}$ against indicated horsepower (fig. 7) and K by direct substitution of the previously determined values in equation (9). Thus for the cylinder tested, the correlating equation can be written

$$\frac{T_v - T_a}{T_g - T_v} = 0.078\ I^{0.48} \left[\frac{1}{(\sigma \Delta p)^{0.24}} + C\right]$$

A curve showing variation in mean effective gas temperature T_g with variation in fuel-air ratio is shown in figure 8.

Influence of Operating Variables on Mean Effective Gas Temperature

After the constants of equation (9) have been evaluated for a given cylinder, it becomes possible to predict the effects of several variables, principally fuel-air ratio, spark timing, and combustion-air temperature on exhaust-valve temperatures in the cylinder by determining their influence on the mean effective gas temperature of the valve. In reference 2 it is shown that the exhaust-valve temperature is greatly influenced by variations in fuel-air ratio. The reason for this effect of fuel-air ratio on exhaust-valve temperature is made apparent by figure 8, which shows the effect of fuel-air ratio on the mean effective gas temperature for the exhaust valve. Small changes in fuel-air ratio result in appreciable changes in T_g.

The influence of other engine variables on valve temperature is less marked. Figure 9 shows an indication of the influence of spark timing on the mean effective gas temperature T_g for the exhaust valve. It is apparent from these data that operating temperatures of the valve are not appreciably affected over the normal range of spark timing. Enough data are not yet available to determine a correlation between combustion-air temperature and mean effective gas temperature. The small amount of data available, however, indicate that a change in combustion-air temperature results in approximately the same change in mean effective gas temperature T_g.

Thermal-Resistance Factor

The thermal-resistance factor C is a measure of the capacity of a valve to transmit heat to the cylinder head through the valve stem and face and through the valve-guide boss. In a given cylinder, once the heat-flow paths from the valve to the surrounding cooling air are fixed by exhaust-port design, the thermal-resistance factor becomes a function of valve design. Therefore, the thermal-resistance factor provides a direct means for evaluating the relative merit of different valve designs and the effectiveness of internal valve coolants on valve cooling. The values of C for the valves tested at an engine speed of 2200 rpm are given in the following table:

Valve	Thermal-resistance factor, C
A	0.726
B	1.315
B (without sodium)	3.54
C	1.023

The thermal-resistance factor of a sodium-cooled exhaust valve, however, may vary with engine speed if insufficient area is provided for the flow of sodium between the head and stem of the valve. The data of figure 10 indicate the value of C for valves A and C is practically independent of engine speed. No data showing the effect of speed on the thermal-resistance factor are available for valve B. Engine speed might have a very marked influence on the thermal-resistance factor of a valve of this design because of the restriction at the throat to the flow of sodium coolant (fig. 4).

The importance of allowing sufficient internal cross-sectional area to permit unrestricted flow of internal coolant is even more apparent from a comparison of thermal-resistance factors for valves B and C. When the internal diameter of the valve was increased at the restricted throat section, the thermal-resistance factor of that valve was reduced to a value approximately 22 percent lower than that of an otherwise similar valve. In a check test, a corresponding reduction in valve temperature of 100° F was observed as shown in figure 11.

Accuracy of Correlating Equation

A comparison of calculated and experimentally determined valve-operating temperatures is shown in figure 12. The calculated values were determined for a cooling-air temperature of 80° F, whereas in the actual tests cooling-air temperature varied between 60° F and 80° F. Figures 11 and 12 indicate that operating temperatures of the exhaust valve in a given cylinder can be predicted with reasonable accuracy provided the constants for the cylinder are known.

SUMMARY OF RESULTS

From an investigation of the operating temperatures of the exhaust valve in a large air-cooled cylinder, the following results were obtained:

1. A semiempirical equation, developed herein, can be used to correlate exhaust-valve temperatures with engine operating conditions and valve design. The general form of this equation is

$$\frac{T_v - T_a}{T_g - T_v} = K\,I^n \left[\frac{1}{(\sigma\Delta p)^m} + C\right]$$

where

T_a	temperature of cooling air
T_g	mean effective gas temperature
T_v	temperature of exhaust-valve crown
I	indicated horsepower
Δp	cooling-air pressure drop across cylinder
σ	ratio of air density at test conditions to air density at sea level
C	thermal-resistance factor, dependent on valve design
K, n, m	constants depending on cylinder and baffle design

2. For the cylinder tested, the values of the cylinder constants were found to be: K, 0.078; n, 0.48; and m, 0.24.

3. The mean effective gas temperature T_g determined for the exhaust valve was found to be extremely sensitive to changes in fuel-air ratio, ranging from 2270° F at a fuel-air ratio of 0.067 to 1700° F at a fuel-air ratio of 0.11.

4. The thermal-resistance factor C in the correlating equation evaluates the thermal resistance from the exhaust-valve crown to the cooling air. Exhaust-valve design and exhaust-valve boss design can be investigated as to their effect on cooling of the exhaust valve by comparing thermal-resistance factors.

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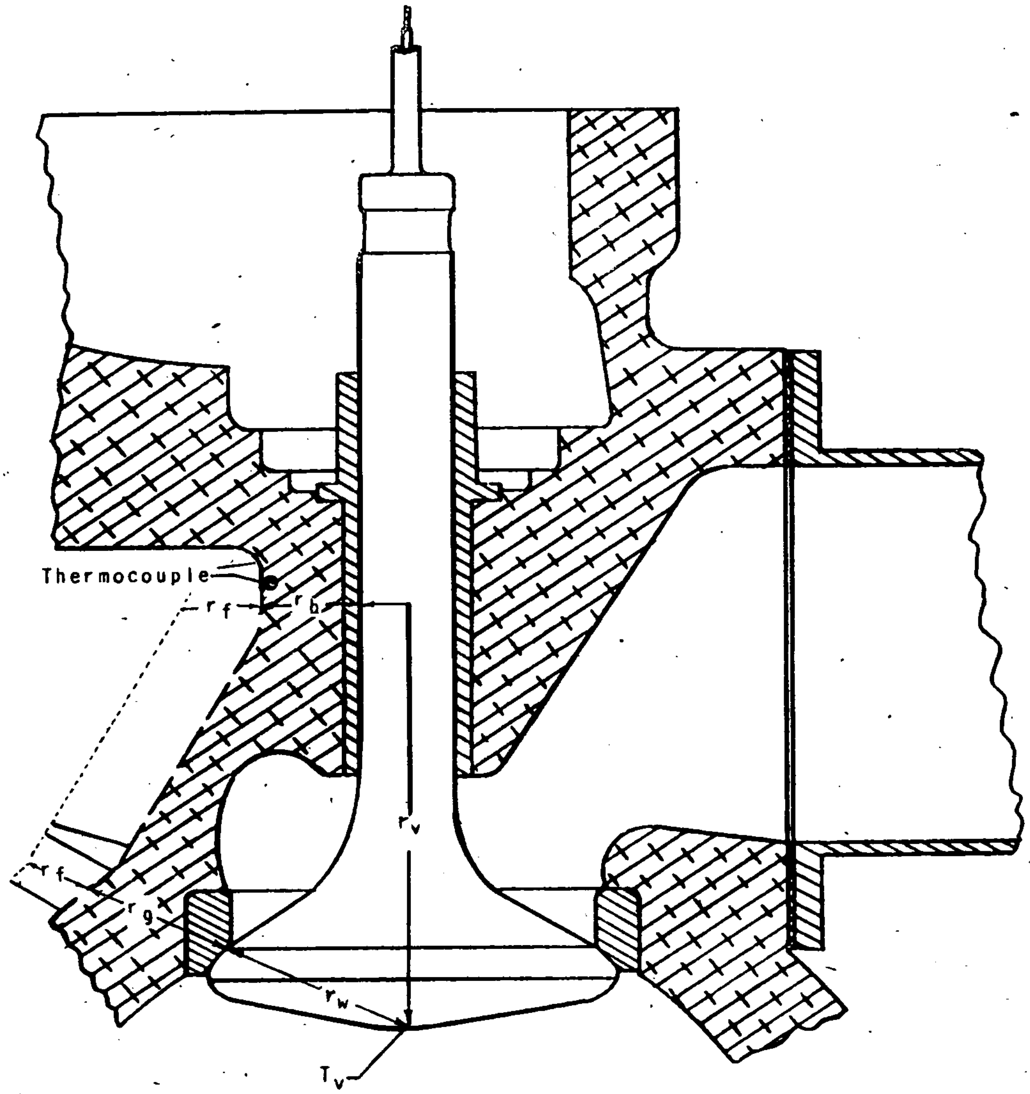
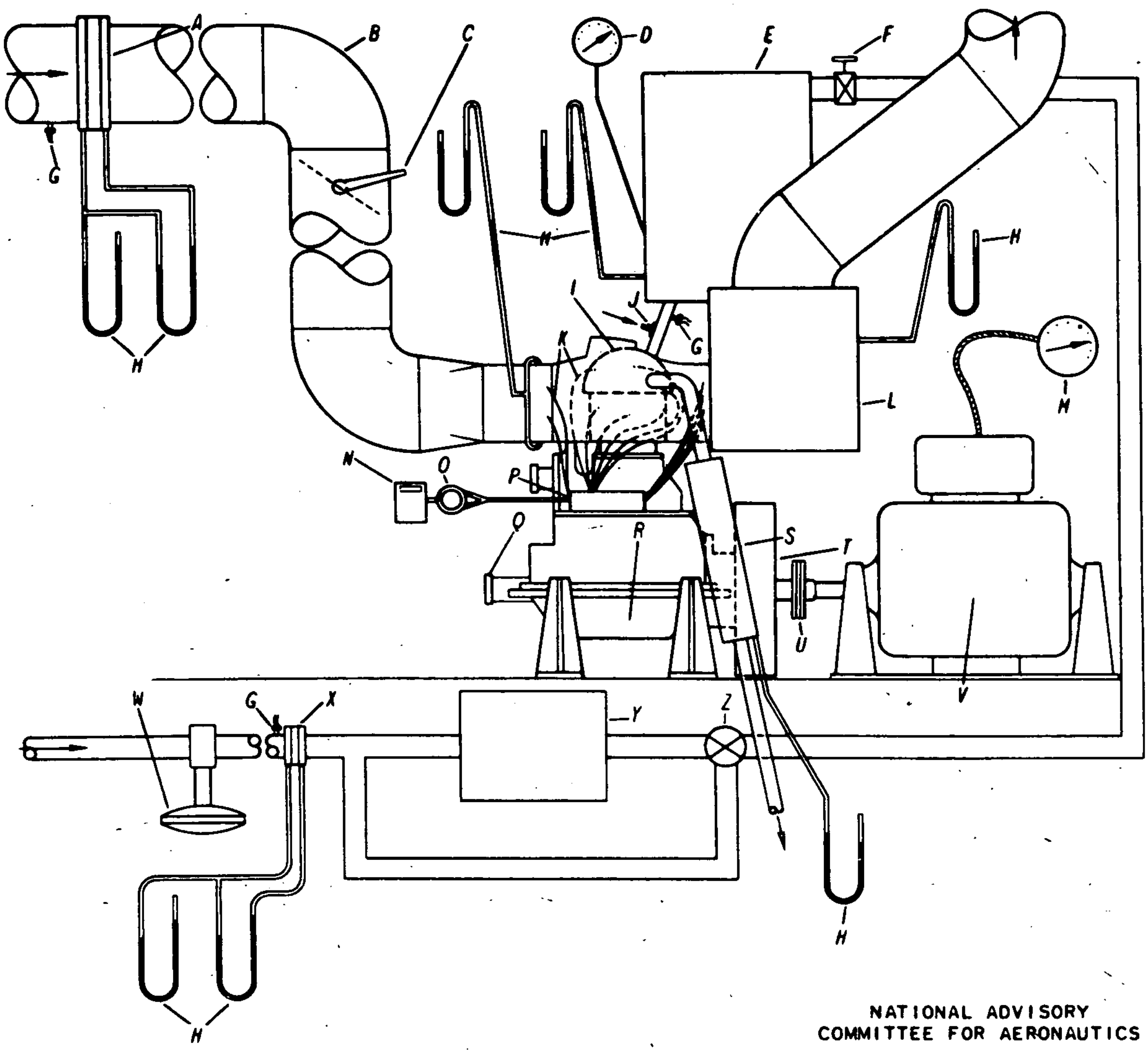


Figure 1. - Thermal resistances between crown of exhaust valve and exterior of cylinder head. Location of thermocouple on outside surface of cylinder head at point nearest exhaust-valve guide.



A Cooling-air orifice
B Cooling-air duct
C Cooling-air pressure regulator
D Remote-reading thermometer
E Surge tank
F Throttle
G Thermocouples
H Manometers
I Test cylinder
J Manifold fuel injector
K Thermocouple leads
L Downstream cooling-air expansion tank
M Remote-reading dynamometer scale
N Self-balancing potentiometer
O Selector switch
P Thermocouple cold-junction box
Q Distributor
R Crankcase
S Exhaust expansion tank
T Flywheel
U Flexible coupling
V Dynamometer
W Combustion-air pressure regulator
X Combustion-air orifice
Y Air heater
Z Motor-controlled mixing valve

Figure 2. - Test engine, auxiliary equipment, and instruments.

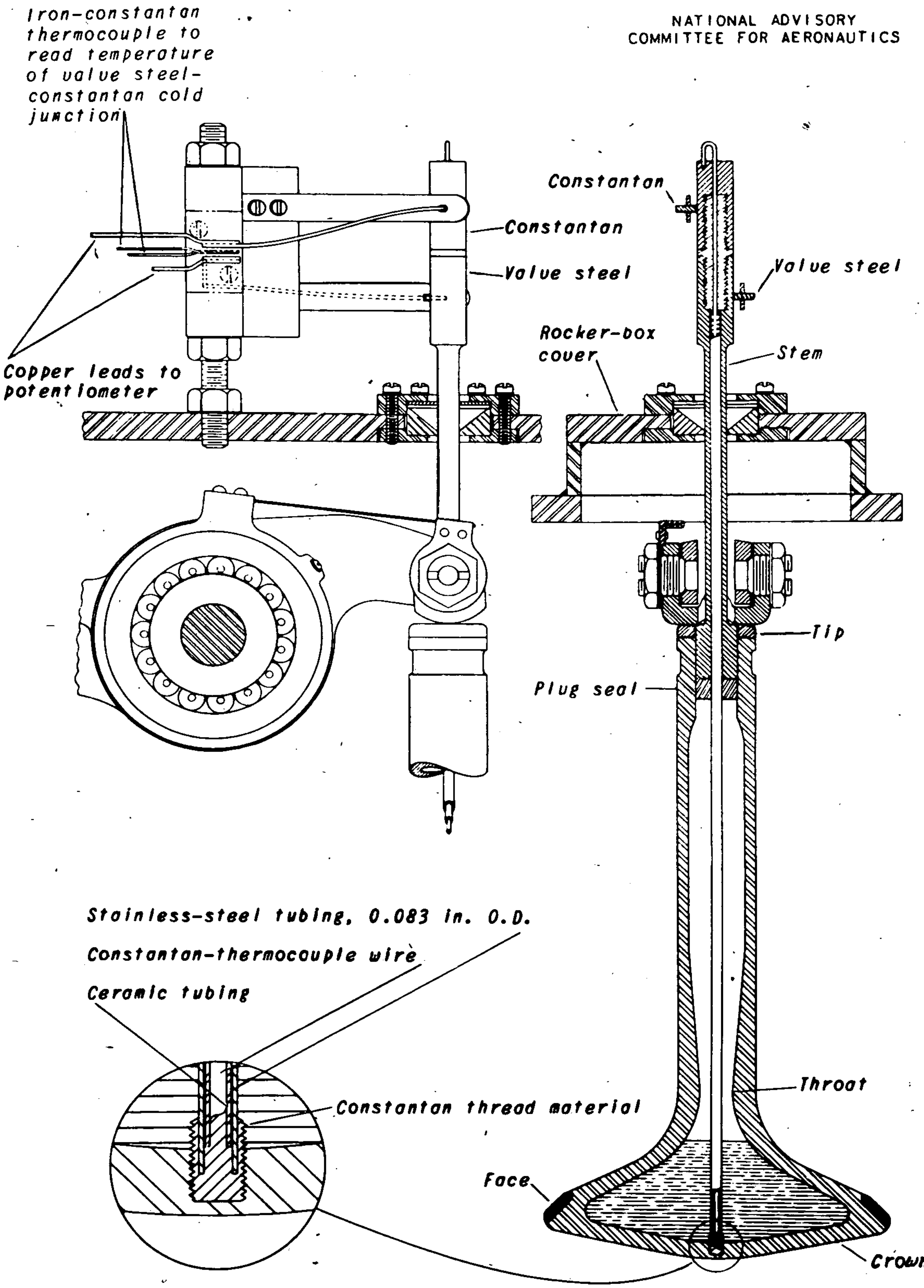


Figure 3. - Details of sodium-cooled exhaust valve equipped with a thermocouple.

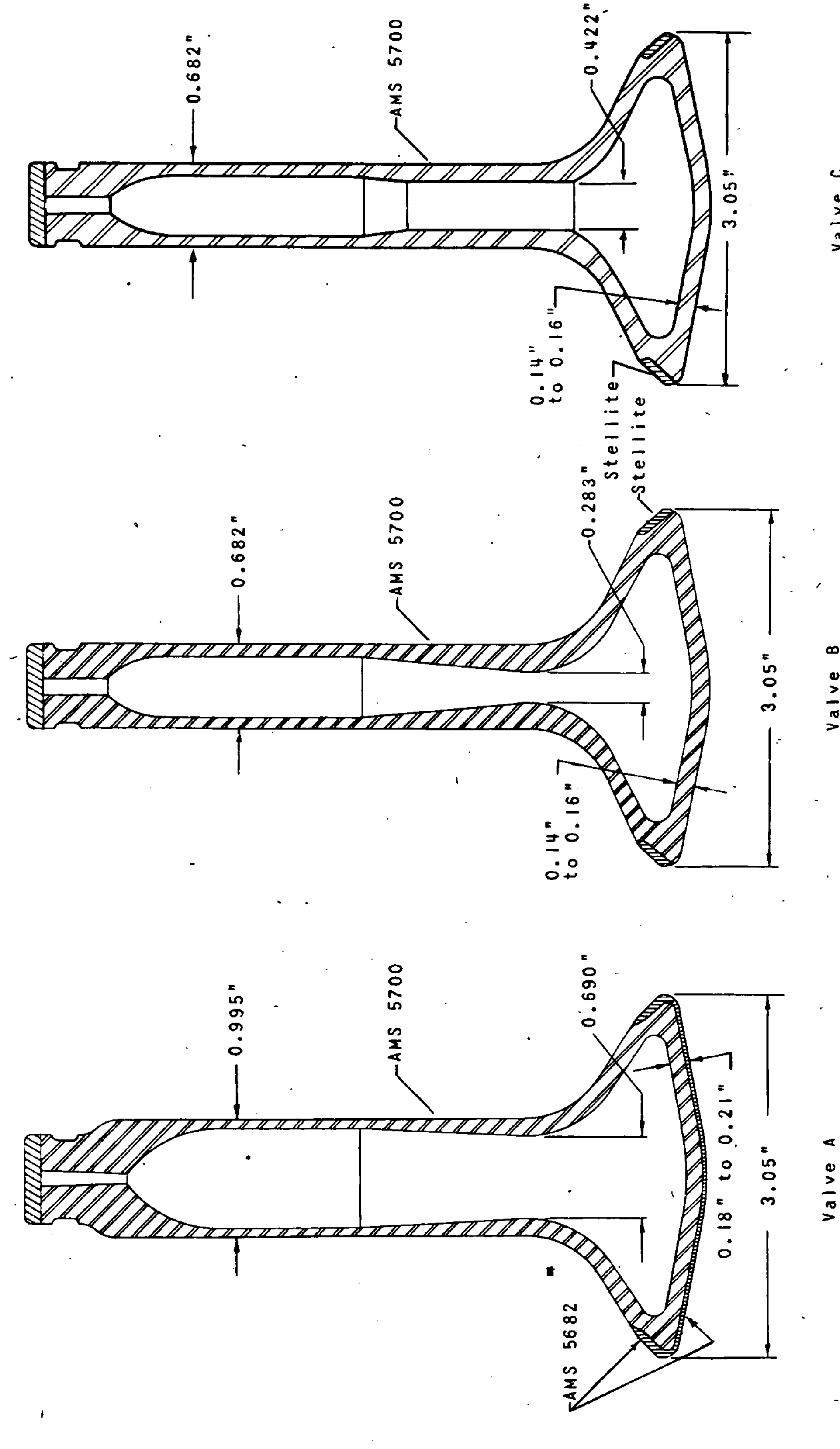


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Figure 4. - Comparison of valve-design features.

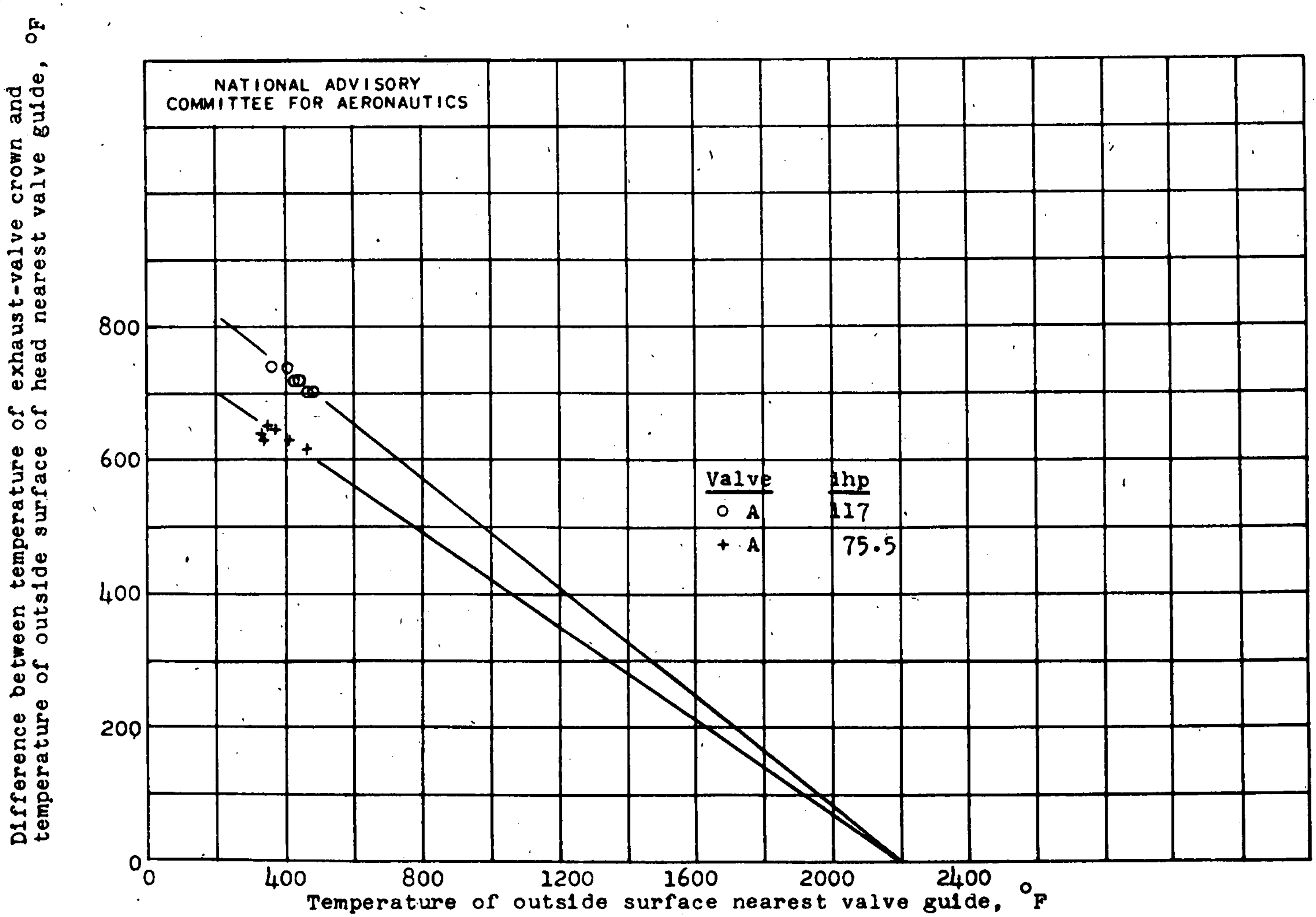

Figure 5. - Determination of effective gas temperature for exhaust valve in the test cylinder. Cylinder displacement, 206 cubic inches; engine speed, 2200 rpm; fuel-air ratio, 0.072; combustion-air temperature, 150° F; spark timing, $22\frac{1}{2}$° B.T.C.

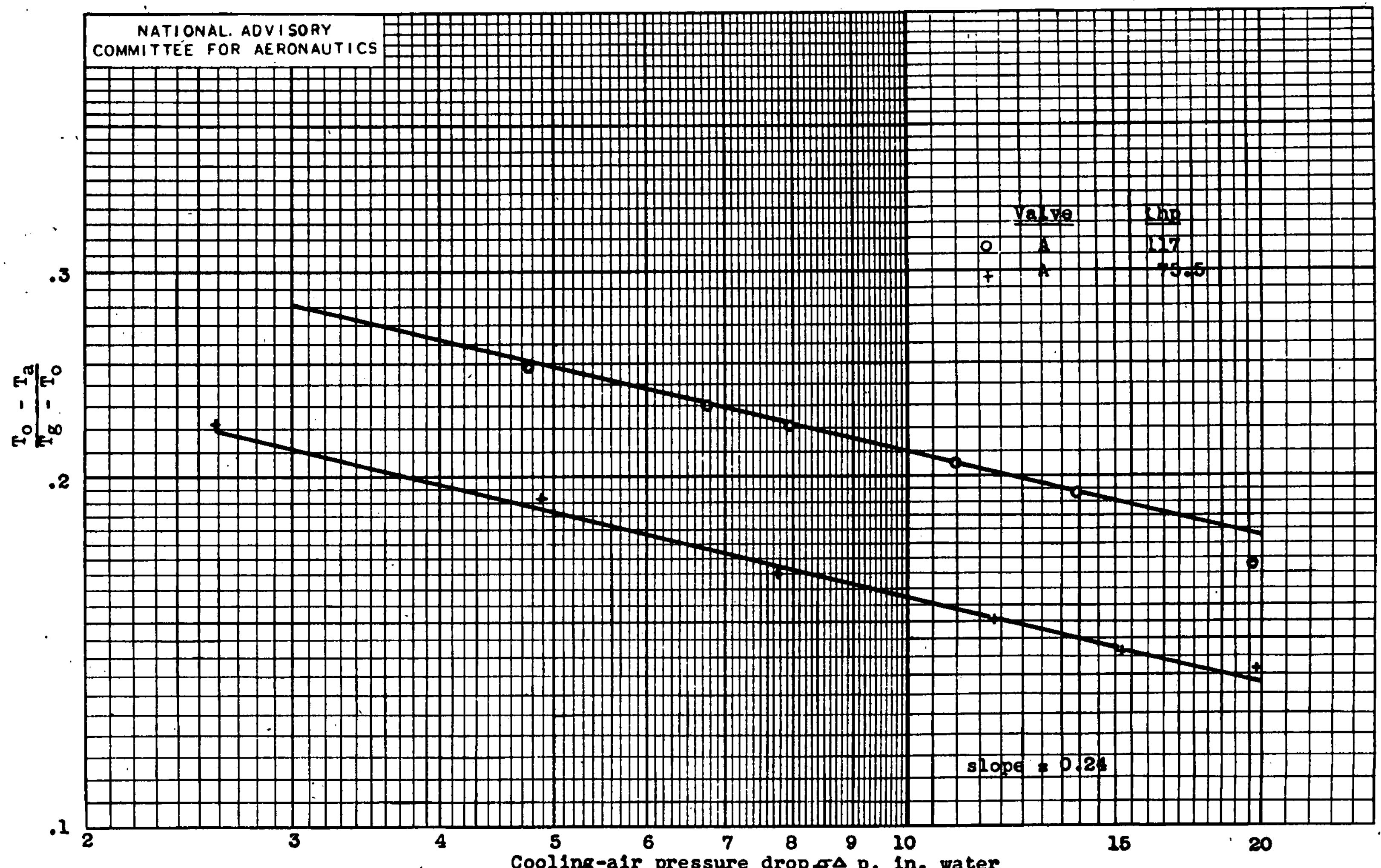

Figure 6.- Variation in $\frac{T_o - T_a}{T_g - T_o}$ with cooling-air pressure drop across cylinder. Cylinder displacement, 206 cubic inches; engine speed, 2200 rpm; fuel-air ratio, 0.072; combustion-air temperature, 150° F; spark timing, $22\frac{1}{2}$ B.T.C.

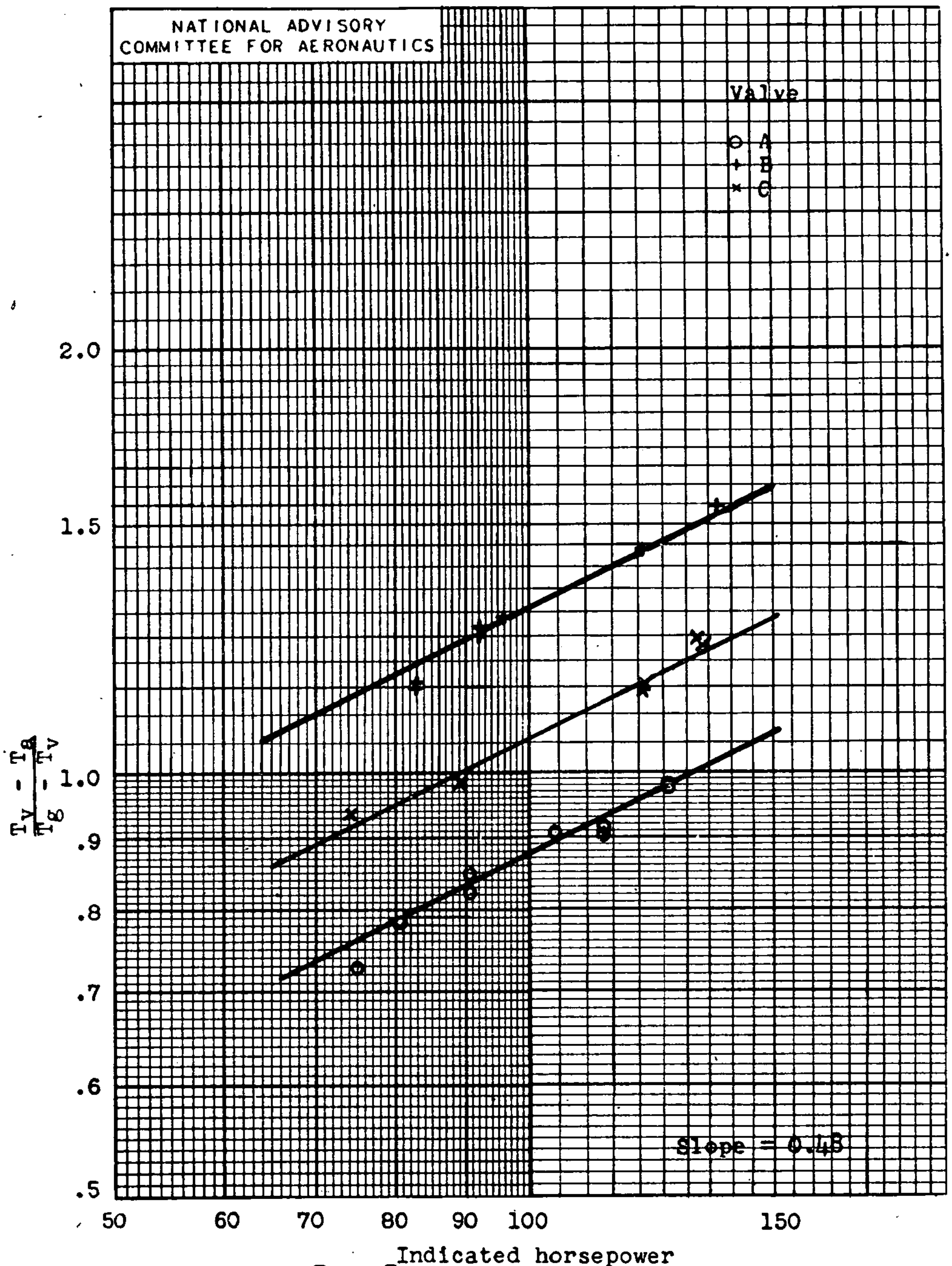

Figure 7. - Variation of $\frac{T_v - T_a}{T_g - T_v}$ with variation of indicated horsepower. Cylinder displacement, 206 cubic inches; engine speed, 2200 rpm; fuel-air ratio, 0.072; cooling-air pressure drop, 16 inches of water; combustion-air temperature, 150° F; spark timing, $22\frac{1}{2}^{\circ}$ B.T.C.

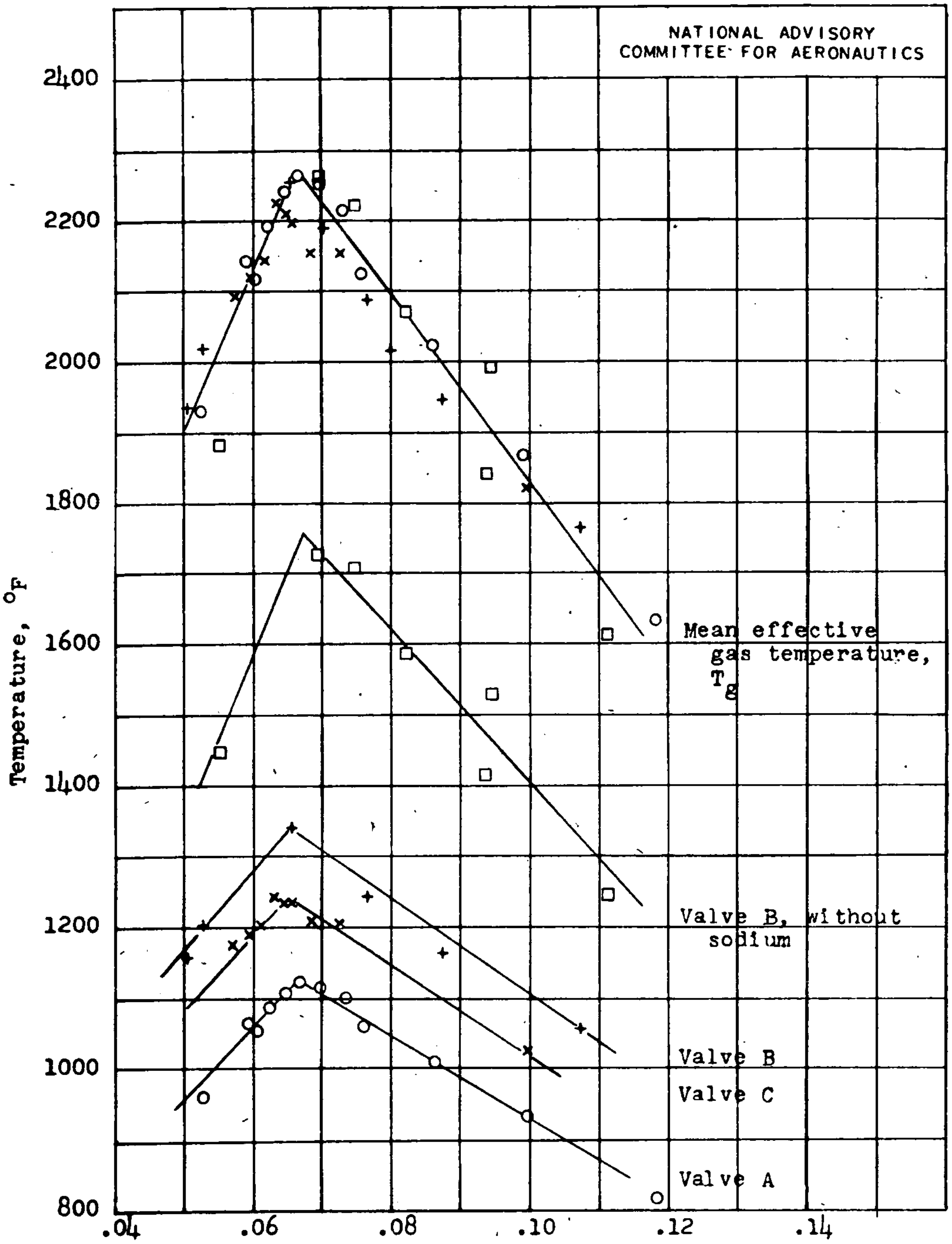

Figure 8. - Variation of mean effective gas temperature and valve temperature with fuel-air ratio. Air-cooled cylinder; cylinder displacement, 206 cubic inches; engine speed, 2200 rpm; indicated horsepower, 117; combustion-air temperature, 150° F; cooling-air pressure drop, 16 inches of water; spark timing, $22\frac{1}{2}$° B.T.C.

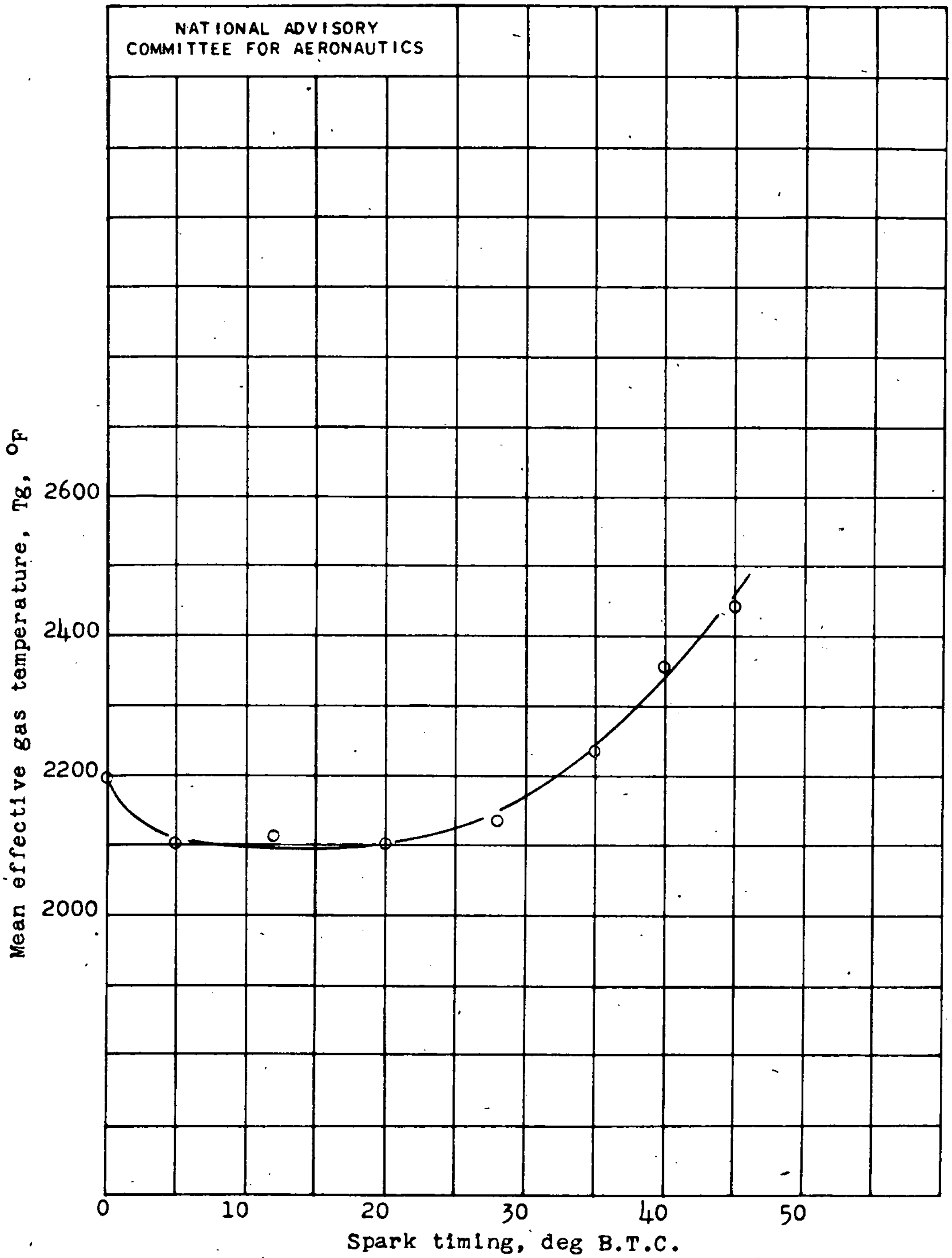

Figure 9. - Effect of spark timing on mean effective gas temperature. Air-cooled cylinder; cylinder displacement, 206 cubic inches; engine speed, 2200 rpm; indicated horsepower, 97; fuel-air ratio, 0.08; cooling-air pressure drop, 16 inches of water; combustion-air temperature, 150° F. (Data from reference 2, fig. 9.)

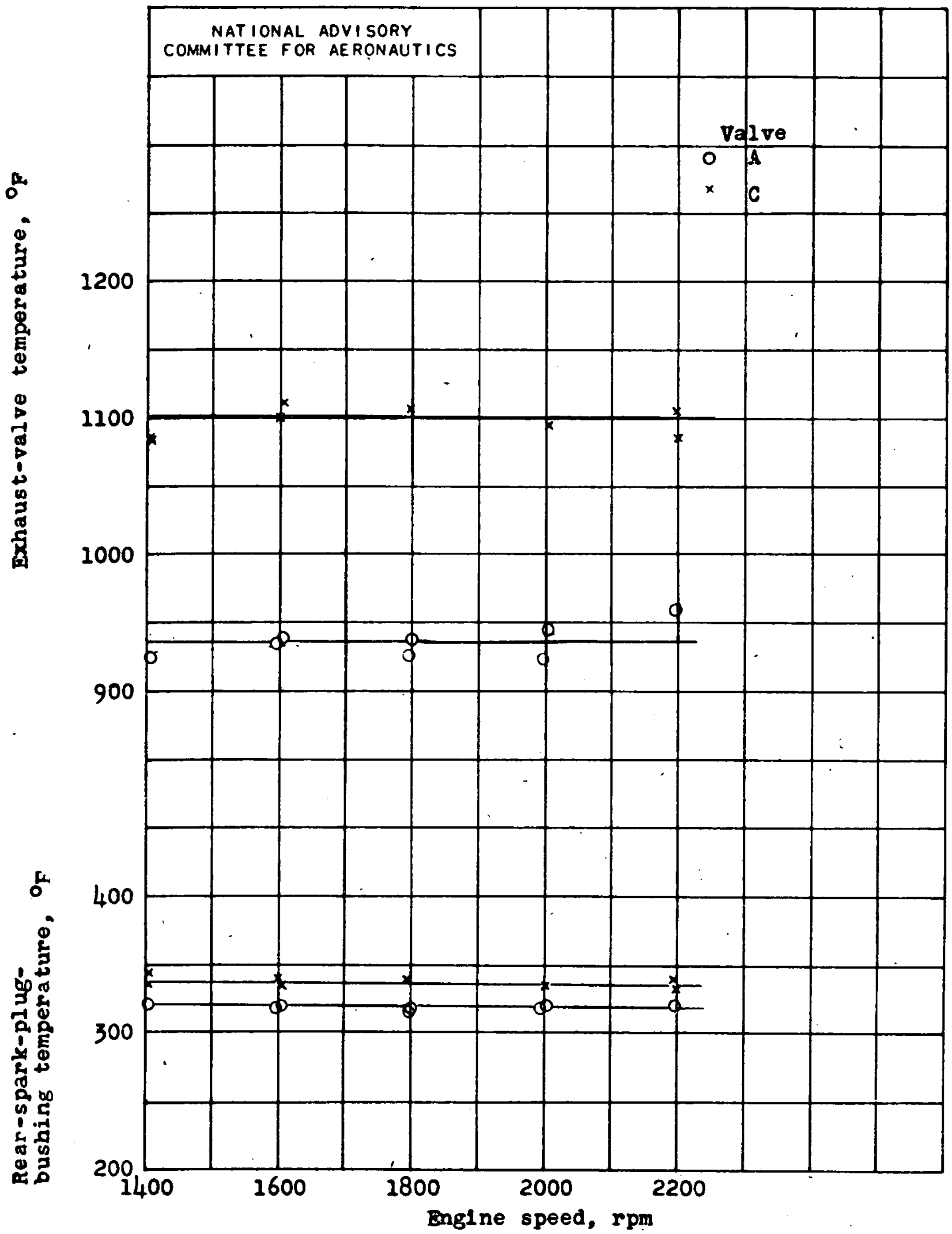

Figure 10. - Variation of exhaust-valve temperature and rear-spark-plug-bushing temperature with engine speed. Air-cooled cylinder; cylinder displacement, 206 cubic inches; indicated horsepower, 71.5; fuel-air ratio, 0.072; combustion-air temperature, 150° F; cooling-air pressure drop, 16 inches of water.

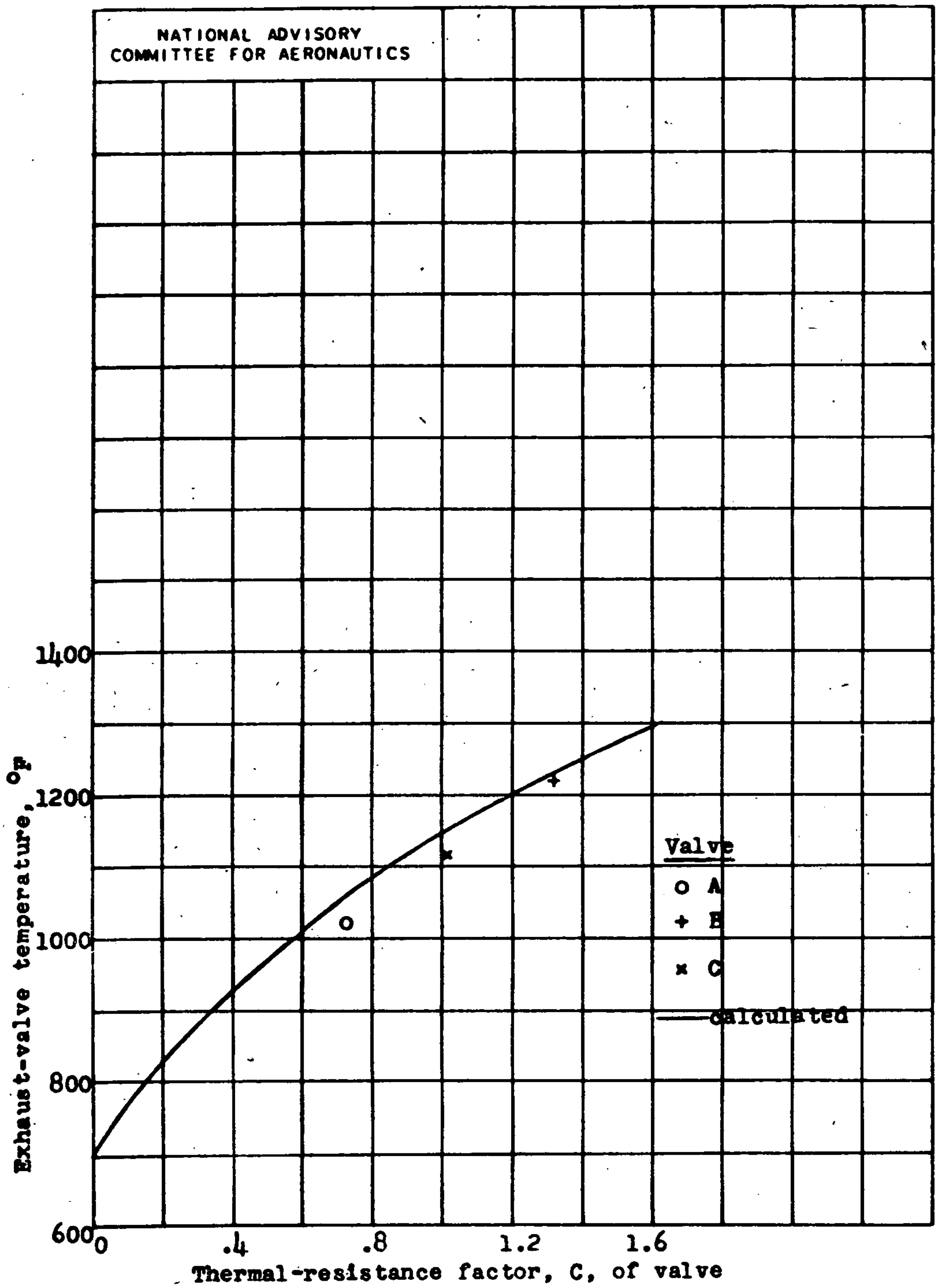

Figure 11. - Effect of the thermal-resistance factor C of three exhaust valves on their temperatures. Air-cooled cylinder; displacement, 206 cubic inches; engine speed, 2300 rpm; indicated mean effective pressure, 215 pounds per square inch; fuel-air ratio, 0.090; combustion-air temperature, 150° F; rear-spark-plug-bushing temperature, 425° F.

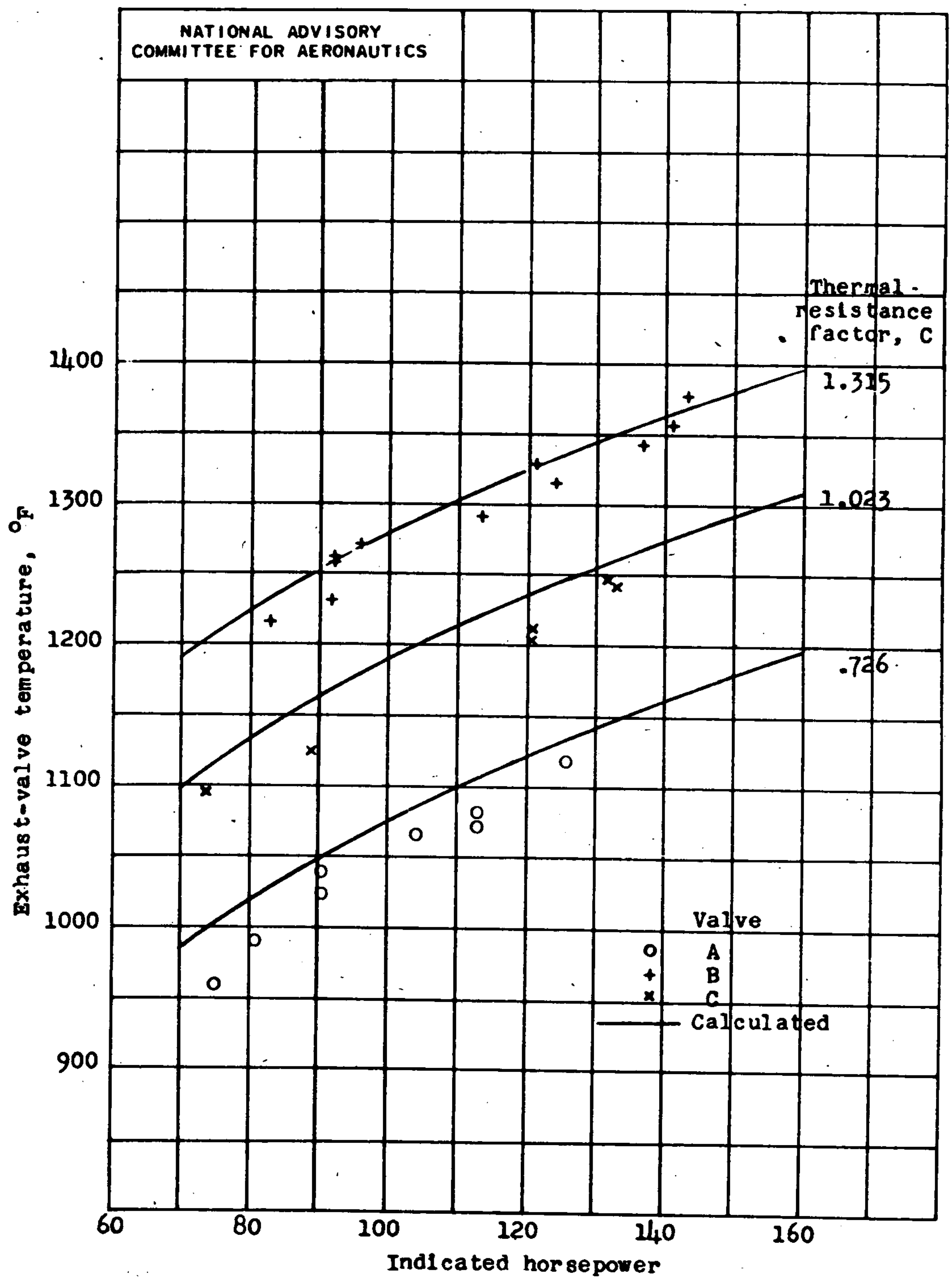

Figure 12. - Comparison of calculated and experimentally determined exhaust-valve temperatures in an air-cooled cylinder. Cylinder displacement, 206 cubic inches; engine speed, 2200 rpm; fuel-air ratio, 0.072; combustion-air temperature, 150° F; cooling-air pressure drop, 16 inches of water.